

# Advanced Numerical and Experimental Techniques for the Extension of a Turbine Mapping

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## ABSTRACT

1D codes are nowadays commonly used to investigate a turbocharged ICE performance, turbo-matching and transient response. The turbocharger is usually described in terms of experimentally derived characteristic maps. The latter are commonly measured using the compressor as a brake for the turbine, under steady “hot gas” tests. This approach causes some drawbacks:

- each iso-speed is commonly limited to a narrow pressure ratio and mass flow rate range, while a wider operating domain is experienced on the engine;
- the turbine thermal conditions realized on the test rig may strongly differ from the coupled-to-engine operation;
- a “conventional” net turbine efficiency is really measured, since it includes the effects of the heat exchange on the compressor side, together with bearing friction and windage losses.

In the present work, advanced experimental techniques aiming to extend the pressure ratio and mass flow rate ranges are summarized and results are compared to conventional test-rig findings.

A recently developed 1D turbine model is described basing on the solution of the flow equations inside the stationary and rotating ducts composing the device. The model is tuned with reference to experimental data collected on the conventional test-rig for a variable geometry turbine (VNT) and a mixed-flow waste-gated turbine (WGT). Then, for the WGT, the model is applied to extend the base map. The results very well agree to the experimental map obtained on the advanced test-rig.

The turbine model is finally used to provide the actual turbine aerodynamic efficiency, which does not account for compressor heat exchange, bearing friction and windage losses. The results are moreover validated against literature derived 3D CFD simulation findings.

The model hence shows the potential to overcome the limitations of a conventional test-rig for a turbine mapping.

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## INTRODUCTION

Current literature is producing more and more numerous contributions dealing with internal combustion engine turbocharging [1,2]. Advanced technologies, such as variable geometry and multi-stage turbocharging, high and low pressure loop EGR systems, assisted turbocharging and turbocompounding, are emerging to control emissions and maintain fuel economy in both diesel and gasoline engines.

More recently, manufacturers are devoting a growing attention to the development of modern turbocharging systems for Spark-Ignition (SI) engines, too. The revival of turbocharged SI engine is based on the need for fuel economy improvement [3,4]. This can be accomplished by reducing the engine size and limiting the throttle usage and pumping losses [5,6]. In order to reach the performance level of larger engines, a turbocharging system is employed, also realizing a

higher low-speed torque, a better vehicle drivability and fun-to-drive (engine downsizing).

During the development of a new turbocharged engine, the computation of the engine and turbocharger matching conditions is a very common task. This is usually realized through the employment of 1D codes [7], which can also provide the boundary conditions for more refined 3D CFD analysis [8]. These procedures require the specification of the turbine and compressor characteristic maps, commonly provided by the manufacturers. In the conventional “hot gas” tests, a compressor is used as a brake for the turbine and, for this reason, a limited operating domain of the turbine map is really available; consequently, various extrapolation techniques of maps' data are commonly included within commercial 1D codes [9,10,11]. The conventional measurement approach, apart from limiting the iso-speed extension of the turbine maps, involves some issues in the aerodynamic turbine efficiency evaluation because of the heat transfer, bearing friction and windage effects.

Due to the above considerations, some advanced techniques are developed at FEV GmbH laboratories aiming to extend the limits of traditional turbine mapping [12,13]. Moreover, a 3D CFD analysis is carried out to investigate the sole aerodynamic performance of the device without accounting for the effects of heat exchange, bearing friction and windage losses. In particular, a conventional test-rig is enhanced thanks to a closed loop circuit on the compressor side (CCL measurements) realizing variable thermodynamic conditions at the compressor inlet. Moreover, replacing the compressor with a bladeless disc, ‘run away’ measurements can be carried out providing the turbine performance at very low mass flow rates and pressure ratios (RA measurements). Finally, using an electric brake instead of a compressor, the low speed operating conditions can be investigated without heat transfer effects on the efficiency evaluation.

During recent years, a 1D turbocharger model has been developed at the DII-ME (Department of Industrial Engineering - Mechanical and Energetic Section) of the University of Naples “Federico II”, with the aim of fitting the experimental data and realizing a physical extension of the performance maps. The simulation procedure characterizes the behavior of small centrifugal compressors [14,15] and centripetal turbines [16] for automotive turbochargers, starting from direct measurements of a reduced set of geometrical data. The model solves the 1D flow equations inside the stationary and rotating ducts included in the devices, and accounts for the main losses occurring in the actual operation. The compressor model, assigning the inner wall temperature and employing proper correlations, provides an estimation of the heat rate. As discussed in the following, the latter is mandatory to properly compare the calculated turbine efficiency with the thermo-mechanical efficiency derived by the experimental tests.

In the next paragraphs, the turbine model is briefly recalled reporting the geometrical procedure and the 1D flow equations. Additional details are reported in [Appendices A](#)

and [B](#). Then, the FEV test bench is described with reference to the maps extrapolation techniques. The 1D turbine model is tuned for two devices, a small waste-gated turbine (WGT) and a variable nozzle turbine (VNT), only basing on the experimental data provided by a conventional test bench. Concerning the first turbine, the procedure is used to extend the map ranges and to predict the aerodynamic performance to be compared to the experimentally derived extended map and to the 3D CFD calculations findings, respectively.

## 1D TURBINE MODEL

The turbine model includes a geometrical module that provides the data required to solve the 1D flow equations inside the different ducts composing the device. In particular, the geometrical schematization includes 5 pipes: the inlet duct, the volute, the nozzle, the wheel and the outlet duct. [Figures 1](#) and [2](#) show the 1D schematization of a WGT and VNT, respectively, while [Figure 3](#) highlights the main stations along the fluid path.

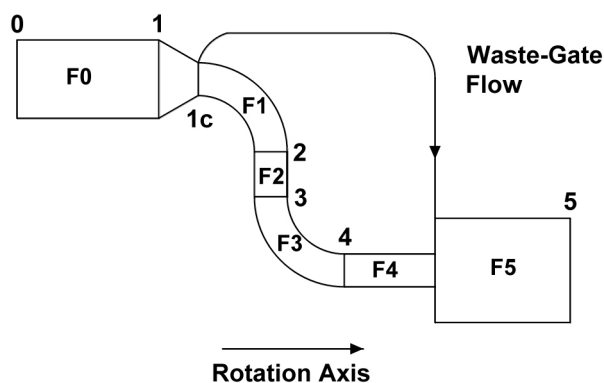


Figure 1. Waste-gate turbine 1D model

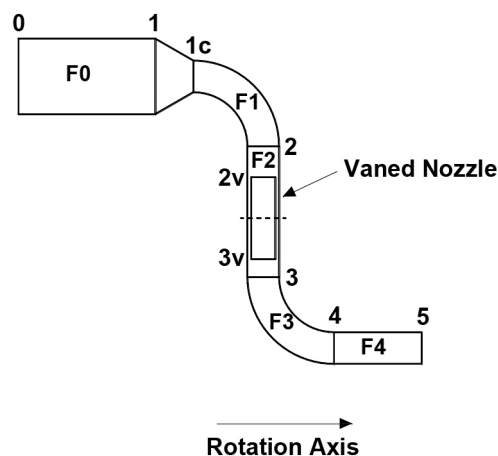


Figure 2. Vaned nozzle turbine 1D model



Figure 3. Turbine section and main stations

## Geometrical Model

The turbine is schematized as follows (Figures 1-2):

- F0 - Inlet pipe (from section 0 to 1) of constant diameter;
- F1 - Volute (from section 1 to 2): it is divided in two parts; the first is a converging cone followed by a linearly variable diameter duct representing the circumferential (1c-2) part of the volute. An equivalent length is considered as half length of the cross section centerline; the outlet cross section area is imposed equal to the nozzle inlet area;
- F2 - Vaned or Vaneless Nozzle (from section 2 to 3): The vaned nozzle is subdivided in 3 different pipes, namely the vaned nozzle itself plus 2 short gaps upstream (2-2v) and downstream (3v-3) the nozzle itself. Its shape is reconstructed once defined the rotation and blade angles at the inlet and the outlet. The vaneless nozzle is considered as a converging pipe of a constant height in the radial direction;
- F3 - Wheel (from section 3 to 4): consisting of  $Z$  rotating pipes,  $Z$  being the number of blades. The profile of the blade-to-blade duct is assigned in terms of blade angles, rotation angles and radii at the inlet and outlet. A numerical procedure is developed to reproduce the 3D wheel geometry basing on the previously listed data. The equivalent 1D profile of blade-to-blade duct (local blade angle, radius, cross section area and wetted perimeter) is automatically provided by the geometrical module, and is employed in 1D flow equations;
- F4 - F5 Outlet pipe (from section 4 to 5) of constant diameter; in case of a WGT, it includes an additional pipe where the by-passed gases mix with the main flow.

A more detailed description of the geometrical module is included in the [appendix A](#).

## 1D Flow Model

### Stationary ducts flow model

The steady 1D flow in the stationary ducts (i.e. the inlet pipe, the volute, the nozzle and the outlet duct) is described by the following equation:

$$[\mathbf{F}]_x = \mathbf{S}_C \quad \mathbf{F} = \begin{Bmatrix} \rho c \\ \rho c^2 + p \\ \rho c h_0 \end{Bmatrix} \quad (1)$$

$$\mathbf{S}_C = - \begin{Bmatrix} \rho c \alpha \\ \rho c^2 \left( \alpha + 2 \frac{f}{d} \frac{c}{|c|} \right) \\ \rho c h_0 \alpha - 4 \frac{q}{d} \end{Bmatrix} \quad (2)$$

The terms  $\rho$ ,  $c$ ,  $p$ ,  $h_0 = c_p T_0 = c_p T + c^2/2$  in the system (1), (2) respectively represent density, absolute velocity, static pressure, total enthalpy per unit mass. The source term,  $\mathbf{S}_C$ , accounts for the duct area variation,  $\alpha = 1/\Omega \, d\Omega/dx$ , the friction coefficient,  $f$ , and the rate of heat exchange,  $q$  (see [appendix B](#)).

### Wheel flow model

Within the wheel, the following balance equations apply:

$$[\mathbf{F}]_s = \mathbf{S}_W \quad \mathbf{F} = \begin{Bmatrix} \rho w \\ \rho w^2 + p \\ \rho w h_{0w} \end{Bmatrix} \quad (3)$$

$$\mathbf{S}_W = - \begin{Bmatrix} \rho w \alpha \\ \rho w^2 \left( \alpha + \frac{2f_w}{d_w} \frac{w}{|w|} \right) + \frac{2f_s}{d_s} \rho c^2 \frac{c_w}{|c|} - \rho u^2 \delta \\ \rho w h_{0w} \alpha - 4 \rho \left( \frac{q_h}{d_h} + \frac{q_s}{d_s} \right) - \rho w u^2 \delta \end{Bmatrix} \quad (4)$$

In the above equations,  $w$  is the relative velocity,  $u = \omega r$ , is the tangential velocity (blade speed) and  $h_{0w} = c_p T_{0w} = c_p T + w^2/2$  is the total enthalpy in the relative motion. The source term  $\mathbf{S}_W$  includes additional contributions arising as a consequence of pipe rotation. They are related to the centrifugal forces acting on the fluid particle and are computed as a function of the radius variation along a streamline:  $\delta = 1/r \, dr/ds$ . Friction forces are subdivided in two contributions, the first related to flow interaction with the impeller surface and the second concerning friction with the case inner surface. Also heat exchange is subdivided in two terms, the first depending on impeller/flow interaction and the second arising from heat exchange on the case.

### Numerical integration of flow equations

Computation starts with a fixed value of mass flow rate at the turbine inlet and imposing inlet total pressure and temperature derived by experimental reference data. Flow

equations are discretized in finite differences and integrated by an iterative procedure.

### Boundary conditions

The boundary conditions in the nodes of adjacent ducts are specified by applying a classical quasi-steady pipe-to-pipe junction problem. Mass and energy conservation at the junction are imposed, while the momentum equation is substituted by a total pressure loss relation between adjacent sections of the connected pipes. Formally, the total pressure loss can be specified as a function of a Mach number expressing the total head loss, as:

$$P_{0ex} = P_{0in} \left( 1 - \frac{\gamma-1}{2} M_{loss}^2 \right)^{\frac{\gamma}{\gamma-1}} \quad (5)$$

Loss related Mach number is opportunely specified according to the junction and the considered loss mechanisms. The model accounts for all the main losses occurring in the turbine: incidence, leakage and blade loading in the wheel and volute losses. Due to the limited generality of employed loss correlation, complexity of the phenomena, and intrinsic limitations of the 1D approach, each correlation includes a tuning constant, which must be identified through comparisons with experimental data. A brief description of various loss mechanisms is reported in the [appendix B](#).

## TURBINE EFFICIENCY

The aerodynamic efficiency of a turbine is defined as:

$$\eta_{turb,aero} = \frac{P_{turb}}{P_{turb,is}} = \frac{\dot{m}_{turb} \Delta h_{0,turb} - \dot{Q}_{turb}}{\dot{m}_{turb} T_{03} \left( 1 - 1/\pi_{ts}^{\frac{k-1}{k}} \right)} \quad (6)$$

being  $P_{turb}$  and  $P_{turb,is}$  the mechanical power provided by the fluid to the turbine and the adiabatic isentropic turbine power, respectively. Due to difficulties in accurately measuring the temperatures downstream and upstream the turbine ( $\Delta h_{0,turb}$ )

and in isolating the heat exchange contribution ( $\dot{Q}_{turb}$ ), a “conventional” net turbine efficiency is usually evaluated as a function of the total enthalpy increase on the compressor side:

$$\eta_{turb,net} = \frac{P_{comp}}{P_{turb,is}} = \frac{\dot{m}_{comp} \Delta h_{0,comp}}{\dot{m}_{turb} T_{03} \left( 1 - 1/\pi_{ts}^{\frac{k-1}{k}} \right)} \quad (7)$$

Imposing the energy balance on the compressor,  $\dot{m}_{comp} \Delta h_{0,comp} = P_{comp} + \dot{Q}_{comp}$ , the mechanical equilibrium equation between turbine and compressor, as schematically shown in [Figure 4](#), results:

$$P_{comp} = P_{turb} - P_{fric} - P_{wind} \quad (8)$$

being  $P_{comp}$ ,  $P_{fric}$ , and  $P_{wind}$  the mechanical power absorbed by the compressor, the bearings friction power and the windage losses, respectively. Since the compressor is not fully adiabatic, the total enthalpy is also affected by the heat transfer phenomena ( $\dot{Q}_{comp}$ ), substantially altering the  $\eta_{turb,net}$  especially at low speed and low load [17, 18, 19, 20, 21, 22, 23].

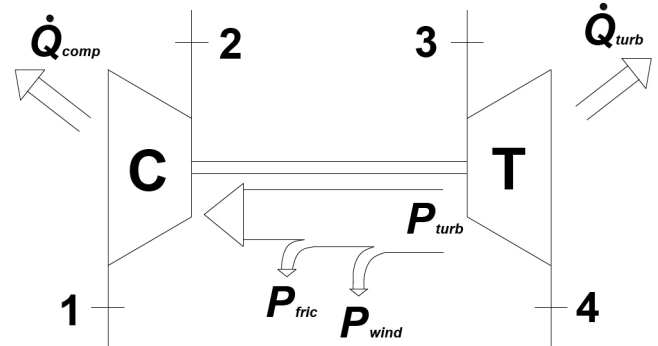


Figure 4. Power and heat flux in a turbocharger

[Figure 5](#) shows a heat transfer map normalized with respect the absorbed mechanical power at different corrected speeds. It is numerically obtained by the compressor model described in [14]. A positive value indicates that the heat enters the fluid. The higher levels are reached at lower speeds and mass flow rates; as previously emphasized, this strongly affects the net turbine efficiency evaluation in the related operating conditions.

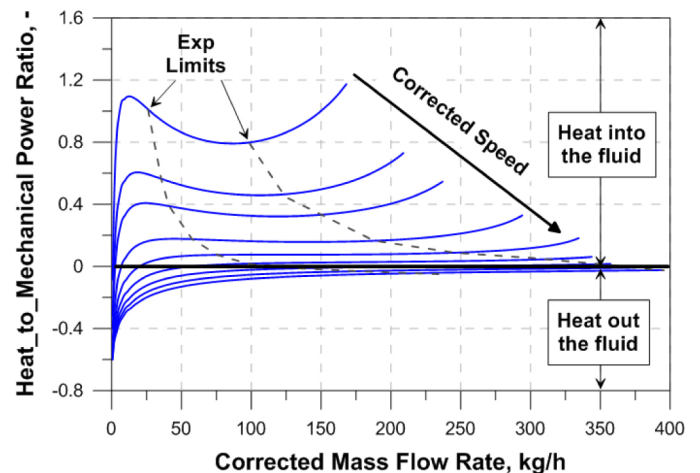


Figure 5. Compressor heat power map

An alternative approach involves the employment of a dynamometer as a brake for the turbine; in this case, a thermo-mechanical efficiency can be evaluated as:

$$\eta_{turb,thermo-mech} = \frac{P_{dynamometer}}{P_{turb,is}} = \frac{P_{turb} - P_{fric} - P_{wind}}{P_{turb,is}} \quad (9)$$

$P_{dynamometer}$  being the power absorbed by the electrical brake. The thermo-mechanical efficiency differs from net power due to the contribution of the heat transfer on the compressor side.

## TEST BENCH FOR TURBINE MAP EXTENSION

This chapter presents several experimental techniques which were developed to extend the turbine mapping range with respect a standard hot gas test bench. Figure 6 shows a “Compressor Closed Loop” test bench. The air flow leaving the compressor is fed back to its inlet. The compressor mass flow rate is controlled using a back pressure valve (BPV). The hot air at the compressor outlet has to be cooled down to the desired inlet temperature. This is achieved by using an air-water heat exchanger with water flow rate regulation. The mass circulating in the loop can be controlled by a blow-off valve on the high pressure side of the back pressure valve and a inflation valve on the low pressure side. At constant compressor pressure ratio and TC speed, inflation increases the compressor mass flow and thus compressor power as shown by Pischinger and Wünsche [24]. To retain the steady state power equilibrium, eq. (8), the turbine power is increased and the turbine pressure ratio can be extended beyond the values reached in a standard hot gas test bench operation (ambient compressor inlet pressure). The air blow-off indeed acts inversely: the turbine pressure ratio can be dropped below the open loop operation.

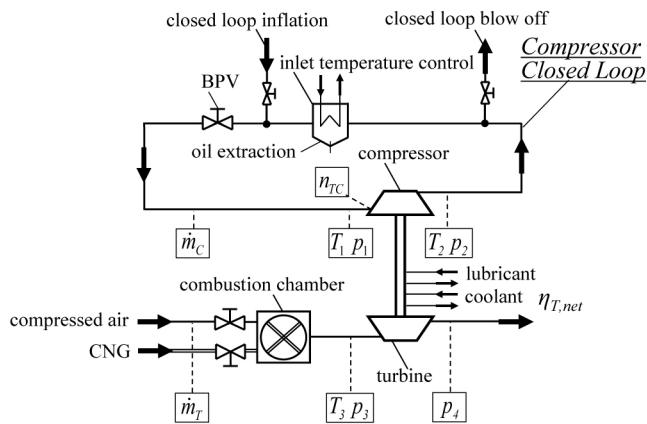


Figure 6. Compressor closed loop test set up for turbine mapping range extension at constant speed

The compressor closed loop approach can usually extend the turbine operating range up to the values corresponding to the peak pressure pulse during an engine exhaust blow-down phase. This is not the case indeed at the pressure pulse minimum, and consequently a further reduction of compressor and turbine power at the test bench is needed.

This is achieved by removing the compressor housing and replacing the compressor wheel with a bladeless disc as shown in Figure 7.

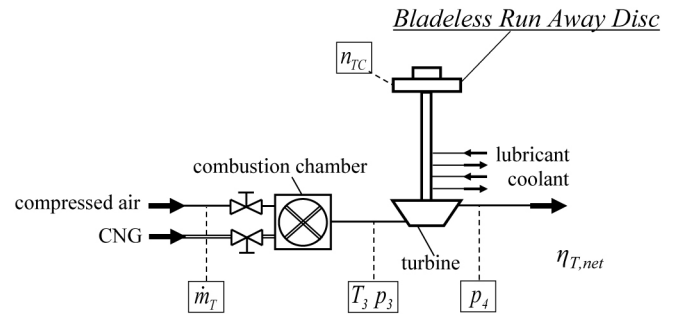


Figure 7. Run away test set up for turbine mapping extension up to zero net efficiency

In this way, the power transferred to the compressor is null and the turbine only delivers the power required to overcome friction and windage losses. Standard measurement on the turbine side allows for the calculation of the turbine performance parameters corresponding to operating conditions of zero net turbine efficiency:

$$P_{comp} = P_{turb} - P_{fric} - P_{wind} = 0 \Rightarrow \eta_{turb,net} = 0 \quad (10)$$

$$P_{turb} = P_{fric} + P_{wind} \Rightarrow \eta_{turb,aero} > 0 \quad (11)$$

A further turbine map extension can be realized, at low TC speeds, directly coupling the turbine to a dynamometer, as shown in Figure 8.

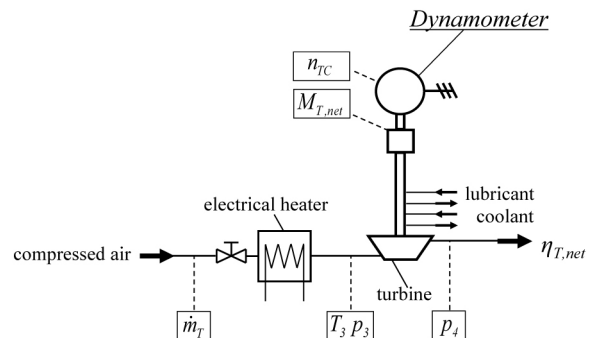


Figure 8. Dynamometer test set up for low speed extension of turbine map

The turbine is driven by pressurized electrically heated gas. At low speeds, in fact, a low turbine power is needed and a combustion chamber for hot gas generation, is in this case not required. The turbine delivered power is absorbed by a speed controlled electrical brake which substitutes the TC compressor. As previously mentioned, this excludes the cause of low speed turbine mapping errors due to heat transferred to the compressor. Turbine power is in fact measured at the